Analysis of the Damping of Sandwich Materials and Effect of the Characteristics of the Constituents

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Abstract

The purpose of the paper is to analyse the damping of sandwich composites made of foam core and laminated skins. Damping parameters are investigated in the case of the bending vibrations of clamped-free beams. Modelling of the damping of the sandwich materials is established using a finite element analysis formulated on the sandwich material theory. Damping modelling is based on the evaluation of the different energies dissipated in the material directions of the core and the layers of the skins. The results obtained in the case of the bending vibrations of beams show a good agreement with the experimental results derived from an experimental investigation. Next, a parametric study is implemented to identify the effect of the characteristics of the constituents on the damping of the sandwich composites.

Keywords

Sandwich Materials; Damping; Finite Element Analysis; Vibration Testing

Introduction

Sandwich composite materials are increasingly being used in a variety of industrial applications such as, marine, aerospace, automobile industry etc. The use of sandwich construction is the result of an increasing demand for light and strong structures. Weight saving is a dominant factor in the transport sectors such as high speed trains, high speed boats, cars etc. The sandwich concept takes advantages over single skin laminated structures in terms of flexural properties with a reduced weight. Flexural stiffness and strength are just two of a variety of design criteria to be considered.

The problem to dissipate energy in structures is an important consideration in mechanical design of structures. Sufficient damping is needed to reduce damping of structures, further to avoid fatigue fracture. Generally, the damping of metal structures is low. For fibre reinforced materials, the damping is higher and depends on the constituent properties. In the vibrations of structures made of sandwich materials, the damping depends on the constituent properties and a high part of the energy is dissipated by the transverse shear effects induced in the sandwich core, which leads to increase the damping of materials.

General considerations on the damping of composite materials are developed in the review of Chandra et al (Chandra, Singh, Gupta, 1999). Firstly, the review considers the different damping processes induced in fibre reinforced composites, resulting from the viscoelastic behaviour of the matrix, the damping of the fibre-matrix interphase, the damping due to damage and the thermo-elastic damping. Next, models for prediction of damping are analysed: the correspondence principle which leads to the introduction of the complex moduli, the stored energy approach, the macro-mechanical description states that the energy dissipated can be considered as the superposition of the energies dissipated in longitudinal, transverse and shear behaviour. At last, the review considers the application of damping models to obtain high damping of laminates combining interleaving viscoelastic layers and fibre arrangement in the different layers of the laminates. Other works on the vibrations of sandwich materials have been reported in the survey papers of Nakra (Nakra, 1981 & 1984). Based on these works, great contributions (Vaswani, Asnani, Nakra, 1988 & Moser, Lumassegger, 1988 & He, Rao, 1992 & Rikards, 1993) were made to the introduction of the concept of a complex modulus, using the elastic-viscoelastic correspondence principle. In this concept, the real part

of the complex modulus represents the elastic stiffness and the imaginary part is associated with the energy dissipation. More recently, the concept of complex modulus was associated with a high-order shear deformation theory by Meunier and Shenoi (Meunier, Shenoi, 2001) to model the damping of sandwich plates. Extensive analyses of the damping of rectangular laminate plates were developed by Berthelot and Sefrani (Berthelot, Sefrani, 2004 & Berthelot, 2006) using the Ritz method that was then extended (Berthelot, 2006 & Berthelot, Sefrani, 2006) to the case of laminates with interleaved viscoelastic layers. The Ritz method is restricted to the analysis of rectangular plates. In the case of complex shape structures, it is necessary to consider a finite element analysis (El Mahi, Assarar, Sefrani, Berthelot, 2008). A finite element was developed by Araújo et al. (Araújo, Mota Soares, Mota Soares, 2010) for the analysis of active sandwich plates with a viscoelastic core and laminated anisotropic face layers, as well as piezoelectric sensor and actuator layers. The elastic layers were modelled with a first order shear deformation theory, and viscoelastic core with a three order shear deformation theory. The dynamic problem is solved in the frequency domain with frequency dependent material properties for the core. Identification of the frequency dependent properties of viscoelastic core materials was considered in (Araújo, Mota Soares, Mota Soares, Herskovits, 2010). Modelling of the damping of sandwich materials has been investigated by Assarar et al. in (Assarar, El Mahi, Berthelot, 2009) using a finite element analysis, and the results derived from which are then applied to the experimental evaluation of the damping parameters of the sandwich materials.

The purpose of the present paper is to consider the modelling of damping of sandwich materials using a finite element analysis based on the sandwich material theory. The results derived from this analysis are then applied to the experimental evaluation of the damping parameters of sandwich materials in the case of the flexural vibrations of beam specimens. At last a parametric study is implemented to identify the influence of the characteristics of the constituents on the damping of sandwich structures.

Modelling the Damping of Sandwich Composite Materials and Structures

To describe the flexural behaviour of sandwich material test specimens, laminate theory or sandwich material theory (Berthelot, 1999 & 2012) can be used, which leads to similar results.

Therefore, the vibrations of the test specimens are investigated in the paper using the QUAD4S finite element type of PERMAS software. This finite element analysis uses the classical sandwich material theory for which the in-plane displacements of the core are linear functions of the z coordinate through the foam core thickness h_c as well the in-plane displacements are uniform through the thicknesses h_s of the skins of the sandwich composite. Moreover, the transverse displacement w is considered as independent of the z coordinate. Thus, the strain ε_{zz} is neglected in the foam core and in the skins.

The finite element analysis gives (Fig. 1) the values of the in-plane stresses σ_{xx}, σ_{yy}, σ_{xy}, in each layer k of the skins of each finite element e of the structure under consideration:

$$\sigma_{xxk}, \sigma_{yyk}, \sigma_{xyk}, \tag{1}$$

and the values of the stresses σ_{xx}, σ_{yy}, σ_{xy}, σ_{xz}, σ_{yz}, on the lower face (l) and the upper face (u) of core of each finite element e of the structure (Fig. 1):

$$\begin{aligned} &\sigma_{xx\mathrm{lc}}, \sigma_{yy\mathrm{lc}}, \sigma_{xy\mathrm{lc}}, \sigma_{xz\mathrm{lc}}, \sigma_{yz\mathrm{lc}}, \\ &\sigma_{xx\mathrm{uc}}, \sigma_{yy\mathrm{uc}}, \sigma_{xy\mathrm{uc}}, \sigma_{xz\mathrm{uc}}, \sigma_{yz\mathrm{uc}}, \end{aligned} \tag{2}$$

with

$$\sigma_{xz\mathrm{uc}} = \sigma_{xz\mathrm{lc}} = \sigma_{xz\mathrm{c}}, \qquad \sigma_{xz\mathrm{uc}} = \sigma_{yz\mathrm{lc}} = \sigma_{yz\mathrm{c}}. \tag{3}$$

Next, the evaluation of the damping is derived from the energy method to perform the evaluation of the different energies evaluated in the material directions as follows.

- In-Plane Strain Energy

  The in-plane energy U_{d}^{e} stored in a given finite element e can be expressed as a function of the in-plane strain energies related to the material directions as:

$$U_{\mathrm{d}}^{e} = U_1^e + U_2^e + U_6^e, \tag{4}$$

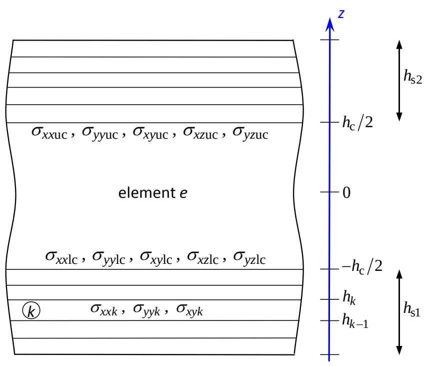


FIG. 1 STRESSES DERIVED FROM FINITE ELEMENT ANALYSIS IN THE LAYERS OF THE SKINS AND IN THE CORE

with

$$U_1^e = \frac{1}{2}\iiint_e \sigma_1 \varepsilon_1 \, dx\,dy\,dz,$$
$$U_2^e = \frac{1}{2}\iiint_e \sigma_2 \varepsilon_2 \, dx\,dy\,dz, \quad (5)$$
$$U_6^e = \frac{1}{2}\iiint_e \sigma_6 \varepsilon_6 \, dx\,dy\,dz,$$

where the integrations are extended over the volume of the finite element e. ε_1, ε_2 , ε_6 and σ_1, σ_2 , σ_6 are respectively the in-plane strains and stresses related to the material directions of the layer k of the skins or the core of sandwich material.

- Transverse Shear Strain Energy

The transverse shear energy for a given finite element e can be expressed in the material directions as:

$$U_d^e = U_{44}^e + U_{55}^e, \quad (6)$$

with

$$U_{44}^e = \frac{1}{2}\iiint_e \sigma_4 \gamma_4 \, dx\,dy\,dz,$$
$$U_{55}^e = \frac{1}{2}\iiint_e \sigma_5 \gamma_5 \, dx\,dy\,dz, \quad (7)$$

where the integrations are extended over the volume of the finite element e. σ_4 and γ_4 are respectively the transverse shear stress and strain in plane (T, T') of material in the layer k of the skins or in the core of sandwich material. σ_5 and γ_5 are the transverse shear stress and strain in plane (L, T'') of the element.

Finally, the damping of the finite element assemblage can be evaluated by extending the energy formulation approach considered in (Berthelot, 2006). The total strain energy stored in the laminated structure is given by:

$$U_d = U_{11} + U_{22} + 2U_{12} + U_{66} + U_{44} + U_{55}, \quad (8)$$

with

$$U_{ij} = \sum_{\text{elements}} U_{ij}, \quad (9)$$

Where U_{ij} the strain energies stored in the finite element assemblage is obtained by summation on the element.

The energy dissipated by damping in the layer k of the skins or in the core of the sandwich material of the element e is derived from the strain energy stored in the layer or in the core as:

$$\Delta U_l^e = \psi_{11l}^e U_{11l}^e + \psi_{22l}^e U_{22l}^e + 2\psi_{12l}^e U_{12l}^e + \psi_{66l}^e U_{66l}^e + \psi_{44l}^e U_{44l}^e + \psi_{55l}^e U_{55l}^e \quad (10)$$

These coefficients are related to the material directions (L, T, T') of the layer k of the skins ($l = k$) or of the core (l = c): ψ_{11l}^e and ψ_{22l}^e are the damping coefficients in traction-compression in the L direction and T direction of the foam, respectively; ψ_{12l}^e is the in-plane coupling coefficient; ψ_{66l}^e is the in-plane shear coefficient; ψ_{44l}^e and ψ_{55l}^e are the transverse shear damping coefficients in planes (T, T') and (L, T'), respectively.

The damping energy dissipated in the element e is next obtained by summation on the core and on the layers of the skins of element e as:

$$\Delta U = \Delta U_c^e + \sum_{k=1}^{n} \Delta U_k^e. \quad (11)$$

Finally, the damping of the finite element assemblage is characterised with the damping coefficient ψ of the assemblage derived from relation:

$$\psi = \frac{\Delta U}{U_d}. \quad (12)$$

The in-plane strain energies U_{11}, U_{22}, $2U_{12}$, U_{66} and the transverse shear strain energies U_{44}, U_{55} related to the material directions, can be expressed as functions of both the in-plane stresses σ_{xx}, σ_{yy}, σ_{xy} and the transverse shear stresses σ_{yz}, σ_{xzy} related to the finite element directions (x, y, z). The formulation of the energies expressed in Eq. (5) and (7) as function of mechanical characteristics of constituents (skins and core) and stresses given by Expression (2) was detailed in the modelling of damping developed Berthelot et al.in (Berthelot, Assarar, Sefrani, El Mahi 2008) and by Assarar et al. in (Assarar, El Mahi, Berthelot, 2009). The approach described in this section as well as the modelling of the damping developed in (Berthelot, Assarar, Sefrani, El Mahi 2008 & Assarar, El Mahi, Berthelot, 2009) allow identifying the effect of characteristics of the constituents particularly the influence of skins on the damping of sandwich structures.

Experimental Procedure

Determination of the Dynamic Properties of the Test Specimens

The damping characteristics of the materials were obtained by subjecting beams to flexural vibrations.

The equipment used is shown in Fig. 2. The test specimen is supported horizontally as a cantilever beam in a clamping block. An impulse hammer, model PCB086 B03, is used to induce the excitation of the flexural vibrations of the beam. A force transducer positioned on the hammer allows us to obtain the excitation signal as a function of the time. The width of the impulse and thus the frequency domain is controlled by the stiffness of the head of the hammer. The beam response is detected by using a laser vibrometer Polytec (OFV 302 R optical head and OFV 3000 conditioner) which measures the velocity of the transverse displacement. Then, the excitation and the response signals are digitalized and processed by a dynamic analyzer of signals developed by Siglab Company. This analyzer associated with a PC computer performs the acquisition of the signals, controls the acquisition conditions (sensibility, frequency range, trigger conditions, etc.), and further operates the analysis of the signals acquired (Fourier transform, frequency response, mode shapes, etc.). In addition, the signals and the associated processings can be saved for post-processing. The system allows the simultaneous acquisition of two signals with a maximum sampling frequency of 50 kHz with a resolution of 13 bits for each channel. The analytical bending response of a clamped-free beam submitted to a concentrated loading has been established in paper (Berthelot, Sefrani, 2004). This analytical response can be applied using the viscous damping modelling or the complex stiffness modelling. Thus, the process of the determination of the dynamic parameters of the test specimens was implemented by fitting the experimental responses with the analytical bending response. The curve fitting was obtained by a least square method by means of the optimisation toolbox of Matlab. The experimental and analytical curve fitting allows us to derive the values of the natural frequencies f_i, and the modal damping coefficient ξ_i (case of damping using viscous damping modelling) or the loss factor η_i (case of damping using the complex stiffness model), related to the specific damping capacity by the relation $\psi_i = 2\pi\eta_i$. The specific damping capacity is usually used to characterise the ratio of the energy dissipated to the energy stored in a structure or an element of structure.

Materials

Sandwich materials were constructed with glass fibre laminates as skins and with PVC closed-cell foams as core. The glass fibre laminates of the skins are cross-ply laminates composed of unidirectional layers of E-glass fibres in an epoxy matrix, arranged in the sequence [0°/90°/90°/0°]. The unidirectional layers were fabricated with unidirectional fabrics of weight 300 gm^{-2} with glass fibres aligned in a single direction.

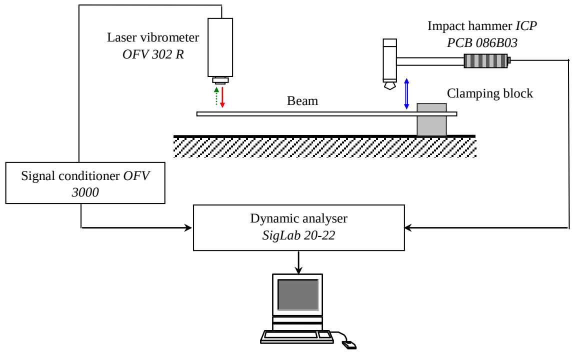


FIG. 2 EXPERIMENTAL EQUIPMENT FOR THE ANALYSIS OF BEAM VIBRATIONS

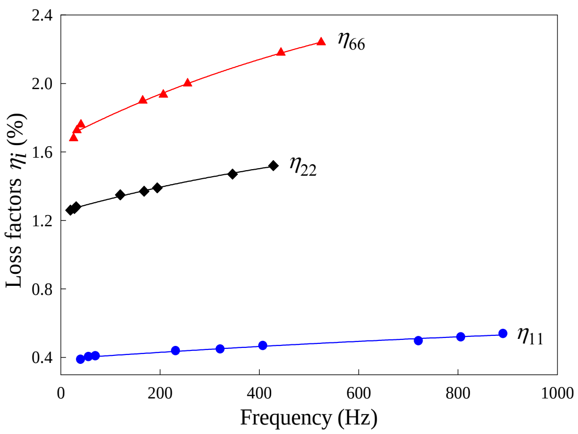
The engineering constants of the unidirectional layers referred to the material directions (L, T, T') or (1, 2, 3) were measured in static tests as mean values of 10 tests for each constant. The values obtained are reported in Table 1. The experimental damping analysis of the laminates has been investigated in (Berthelot, Sefrani, 2006 & Berthelot, Assarar, Sefrani, El Mahi, 2008) and the values of the loss factors derived from this analysis are reported in Fig. 2a. The PVC closed-cell foams were supplied in panels with thickness of 15 mm, and two foams considered differing in their densities: 60 kg m^{-3} and 200 kg m^{-3}.
Mechanical characteristics of the foams were measured in static tensile tests for the Young's modulus and the Poisson's ratio, and in static shear tests for the shear modulus. The results obtained show that the foams are isotropic and the modulus values derived are reported in Table 2. The dynamics properties of the core foam are deduced from the experimental investigation and modelling analysis according to the procedure described in (Assarar, El Mahi, Berthelot, 2012). The values of the dynamics properties deduced from this analysis are reported in Figures 2b and 2c.
Sandwich materials were constructed both with these foams and cross-ply glass-fibre laminates prepared by hand lay-up process, which leads to a nominal thickness of 1.2 mm for the sandwich skins.

TABLE 1. ENGINEERING CONSTANTS OF THE UNIDIRECTIONAL GLASS FIBRE LAYERS.

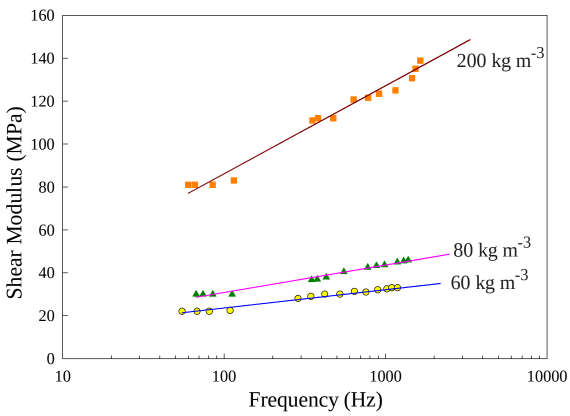
E_L (GPa)	E_T (GPa)	G_{LT} (GPa)	ν_{LT}
29.9	7.5	22.5	0.24

TABLE 2. CHARACTERISTICS OF PVC FOAMS DERIVED FROM STATIC TESTS

Density of the foam (kg m^{-3})	Young's modulus (MPa)	Poisson's ratio	Shear modulus (MPa)
60	59	0.42	22
200	240	0.45	80



(a)



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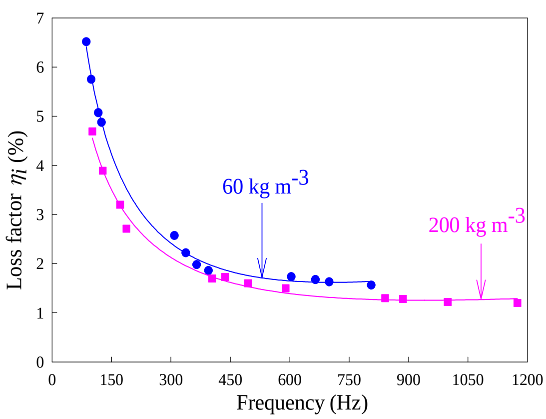


(c)

FIG. 3 DYNAMICS PARAMETERS OF THE CONSTITUENTS OF THE SANDWICH MATERIALS AS A FUNCTION OF FREQUENCY: (a) LOSS FACTORS IN THE MATERIAL DIRECTIONS OF THE UNIDIRECTIONAL GLASS FIBRE LAYERS, (b) SHEAR MODULUS OF FOAMS (c) DAMPING OF FOAMS DEDUCED FROM EXPERIMENTAL INVESTIGATION AND FINITE ELEMENT ANALYSIS.

Characteristics Factors of the Damping of THE Sandwich Materials

Analysis of the Damping of Sandwich Materials

Investigation of the damping has been implemented in the case of sandwich materials considered in previous Subsection B. The experimental investigation was carried out in the case of the flexural vibrations of clamped-free beams. An aluminium spacer was added in the root section which was closely clamped in a rigid fixture. The free length of the beams was equal to 250, 300, 350 and 400 mm, and the beam width was equal to 40 mm.

The experimental results obtained have been compared with the results deduced from the modelling considered in Section II. This analysis takes into account the variation of the damping of the unidirectional layers of the skins (Fig. 3a), as well as the variation of the shear modulus and damping of foam core as functions of the frequency (Fig. 3b and 3c).

Comparison has been made in Fig. 4 between the results deduced from the experimental investigation and from modelling for the sandwich materials, in the case of core with foams of 60 and 200 kg m^{-3}, showing a good agreement between the experimental results and the results deduced from modelling. Furthermore, it is observed that damping as a function of the frequency is discontinuous from one mode to the other, due to a change of the distribution of the strain energies according to the mode. Indeed, the evolution of the damping of the sandwich materials depends on different factors as demonstrated in (Assarar, El Mahi, Berthelot, 2009).

Energies Dissipated in the Core and Skins

Fig. 5 reports the energies dissipated in the core and in the skins for the first four modes of the bending vibrations of the sandwich beams of lengths 250, 300, 350 and 400 mm, in the case of foam cores of density 60 and 200 kg m^{-3}, respectively. These results are derived from a finite element analysis. For the densities of the foam core equal to 60 kg m^{-3}, the energy dissipated in the foam core is clearly higher than that in the skins. Likewise, it is observed an increase of the energy dissipated in the foam core and a decrease of the energy dissipated in the skins when the frequency as well as the mode number increases which can be attributed to the dominant effect of the transverse shear in the foam core, resulting from the low values of the shear modulus of the foam core and

the increment with the mode because of the mode shape. In contrast, for the density of 200 kg m^{-3} of the foam core, the energy dissipated in the foam core increases with the frequency and the mode number, when the energy dissipated in the skins decreases. Moreover, the energy dissipated in the foam core is smaller for the modes 1 and 2 and higher for the mode 4. For this foam density the value of the shear modulus is high, and the results observed can be associated with two different elements: the distribution of the energy in the core and skins according to the modes, and the increase of the shear modulus as a function of the frequency. For the modes 1 and 2, the transverse deformation of the foam core is less pronounced than that for the mode 4, which yields, with a high value of the shear modulus of the foam core, a low energy dissipated in the foam core.

The preceding effect according to the mode shape is also underlined when the evolution of the energies dissipated is taken into consideration as a function of the ratio of the core thickness to the skin thickness (Fig. 6). In this figure, the dissipated energies derived from finite element analysis are reported for the first three modes of the bending vibrations of sandwich beams of 350 mm length and a foam density of 60 kg m^{-3}. Skins of the beams are [0°/90°/90°/0°] cross-ply laminates. It is observed that the energy dissipated in the core increases with the thickness of the foam core. For the first mode, the energy dissipated in the skins is higher than that in the foam core for low values of core thickness: values of core thickness smaller about four times than the skin thicknesses. Additionally, the energy dissipated in the foam core increases when the mode number rises, which shows clearly that the part of the transverse shear energy dissipated in the foam core increases with the mode number. This effect is induced by the shapes of the vibration mode.

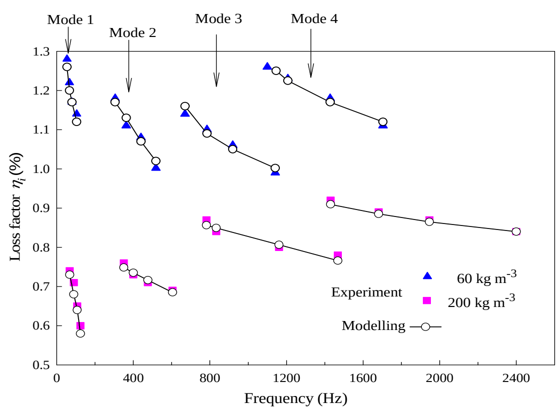


FIG. 4 COMPARISON BETWEEN THE RESULTS DEDUCED FROM EXPERIMENT AND MODELLING FOR THE DAMPING OF SANDWICH MATERIALS, IN THE CASE OF CORE WITH FOAMS OF DENSITY 60 AND 200 kg m^{-3}

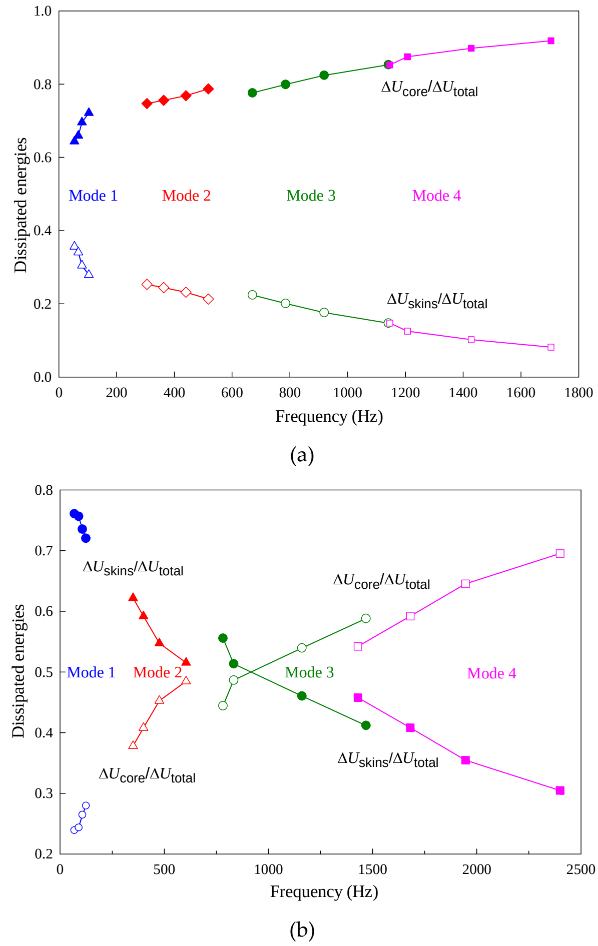


FIG. 5 ENERGIES DISSIPATED IN THE CORE AND IN THE SKINS FOR THE FIRST FOUR MODES OF THE BENDING VIBRATIONS OF THE SANDWICH BEAMS OF LENGTHS 250, 300, 350 AND 400 MM : (a) IN THE CASE OF A FOAM CORE OF DENSITY 60 KG M–3, (b) IN THE CASE OF A FOAM CORE OF DENSITY 200 KG M–3

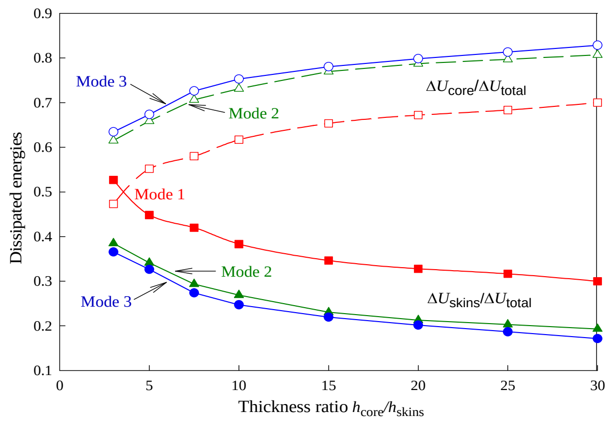


FIG. 6 ENERGIES DISSIPATED IN THE CORE AND IN THE SKINS FOR THE FIRST THIRD MODES OF THE BENDING VIBRATIONS OF SANDWICH BEAMS OF LENGTHS 350 MM AS FUNCTIONS OF THE RATIO OF CORE AND SKINS THICKNESSES, IN THE CASE OF A FOAM CORE OF DENSITY 60 KG M–3 AND [0°/90°/90°/0°] SKINS.

Influence of the Shear Modulus of the Foam Core

For a given damping of the foam core, one of the principal factors which govern the damping of

sandwich materials is the shear modulus of the foam core. Fig. 7 shows the results derived from a finite element analysis for the damping of sandwich materials as a function of the value of the shear modulus of the foam core, in the case of the first two modes of sandwich beams with lengths 250, 300, 350 and 340 mm. The results show clearly that an increase of the shear modulus of the foam core yields a significant decrease of the damping of sandwich materials: when the shear modulus is multiplied by 8, the sandwich damping is divided by a factor about 4 for the first mode and about 3 for the second mode. For a given mode and shear modulus of the foam core, damping of sandwich materials decreases when the frequency increases, then when the length of the sandwich beams is increased. This result can be associated with a decrease in the part of the transverse shear strain energy stored in the foam core. The variation of the damping from one mode to the other is associated with the distribution of the strain energies in the skins and the core. Elements on this aspect are considered in the following subsections.

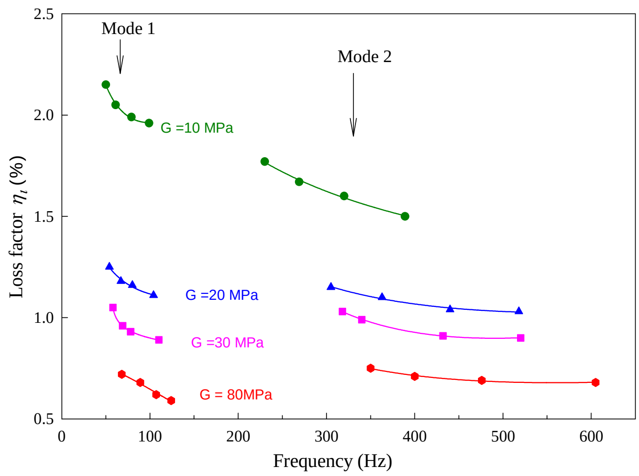
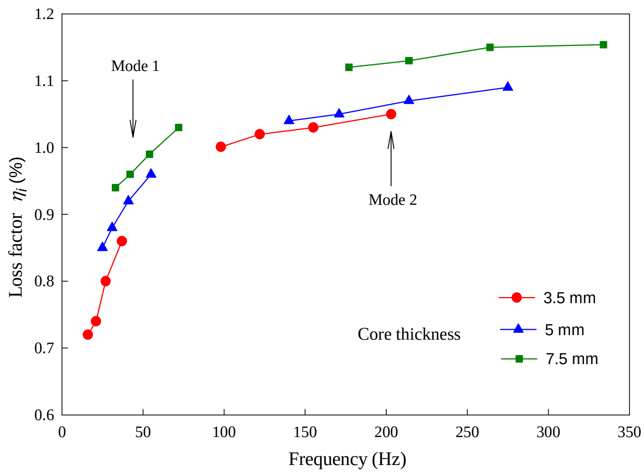


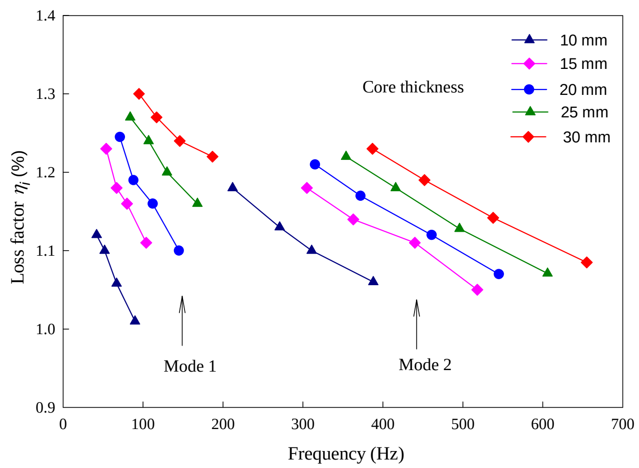
FIG. 7 INFLUENCE OF THE SHEAR MODULUS OF THE FOAM CORE ON THE DAMPING OF SANDWICH MATERIALS, FOR THE FIRST TWO MODES OF SANDWICH BEAMS.

Effect of the Core Thickness on the Damping of Sandwich Materials

Fig. 8 shows the evolution of the damping of the sandwich materials as a function of the thickness of the foam core, for a foam density of 60 kg m^{-3}. Two different evolutions of the loss factors are observed a function of the frequency according to the foam thickness: for thicknesses of the foam core of 3.5, 5 and 7.5 mm, the loss factor increases as a function of the frequency (Fig. 8a), when the loss factor decreases for thicknesses of the core higher or equal to 10 mm (Fig. 8b). These two different behaviours can be associated with the evolution of the distribution of the energies dissipated in the core and skins as a function of the core thickness. For low values of the core thickness, the effect of the damping of the skins is dominant and the increase of the damping of the skin layers with frequency (Fig. 3a) induces an increase of the resultant damping of the sandwich materials. For high values of the skin thicknesses, the effect of the damping of foam core is dominant and the increase of the shear modulus of the foam with the frequency (Fig. 3b) as well as the decrease of the foam damping (Fig. 3c) lead to a decline of the damping of the sandwich materials as a function of the frequency.



(a)



(b)

FIG. 8 EVALUATION OF THE DAMPING OF THE SANDWICH MATERIALS FOR VARIOUS THICKNESSES OF THE FOAM CORE, IN THE CASE OF [0°/90°/90°/0°] CROSS-PLY LAMINATE SKINS AND A FOAM DENSITY OF 60 KG M-3: (a) THICKNESS CORE SMALLER THAN 7.5 MM AND (b) THICKNESS CORE HIGHER THAN 10 MM.

Effect of the Skin Thickness on the Damping of Sandwich Materials

Fig. 9a shows the evolution of the damping of the

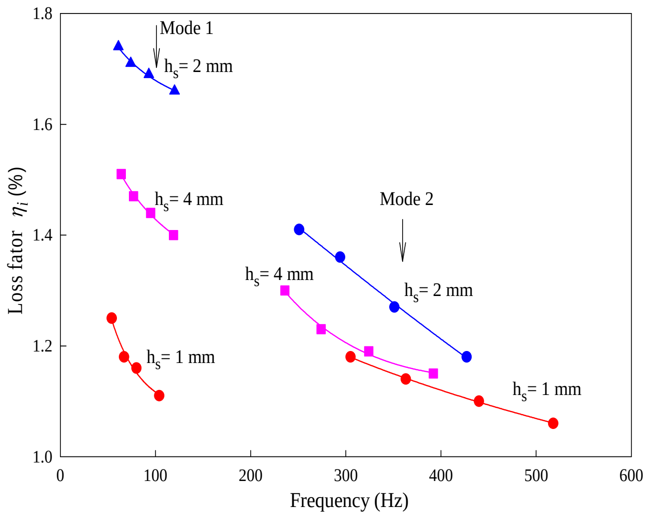
sandwich materials as a function of the thickness of the foam core, for a foam density of 60 kg m-3, in the case of the first two bending modes of sandwich beams with lengths 250, 300, 350 and 400 mm. It is observed that the loss factor decreases as a function of the frequency for each thickness of skins, which results from the evolution of the distribution of the energies dissipated in the core and skins (Fig. 5 and 6). Indeed, the effect of the damping of foam core is dominant as well as the decrease of the foam damping (Fig. 3c), which leads to a decrease of the damping of the sandwich materials as a function of the frequency. Fig. 9b gives the evolution of damping according to the ratio of the skin thicknesses to the core thickness. The damping rises when the ratio of the thicknesses increases with a maximum at about 7.5 which corresponds to a thickness of 2 mm skins. In fact, when the thickness of the skin varies from 1 to 2 mm, the flexural stiffness of skins increases and imposes an important transverse shear stress on the core and consequently an increase of the energies dissipated, leading to an increment of the damping of the sandwich material. When the thickness of skins is higher than 2 mm, flexural stiffness increases but the effect of the thickness of skins on the damping becomes significant. In this case, damping of sandwich materials is dominated by the damping of the skins.

Effect of the Type of Skin on the Damping of Sandwich Materials

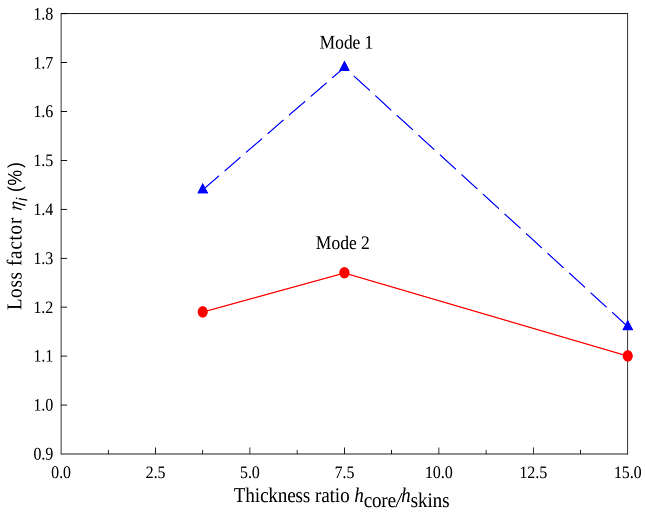
Two materials sandwiches composed of skins with various reinforcements and core foam with density of 60 kg m-3 and 15mm thickness have been considered. The first one is a sandwich materials with [0°/90°/90°/0°]s cross-ply laminates and the second one is a sandwich materials with serge laminate. Fig. 10 compares the results obtained in the case of both sandwich materials, for the first two bending modes. The results show that the damping of serge sandwich is clearly higher than that of cross-ply laminates sandwich. In addition to the dominant effect of the core foam on the damping of the sandwich materials, these results show the effect of the reinforcement type of the skins on the damping of the sandwich materials.

Effect of the Skins Orientations on the Damping of Sandwich Materials

Sandwich materials made of unidirectional fibre skins of 2 mm thickness and foams core of density 60 kg m-3 and 15 mm thick have been in consideration together with various orientations of skins fibres: 0°, 15°, 30°, 45°, 60°, 75° and 90°. Fig. 11a shows the evolution of the damping according to the frequency for the first two bending modes of beams and for different skins orientations. For a given orientation of the fibres, it is observed that damping decreases when the frequency is increased. Skins orientations slightly influence the global damping of sandwich materials in the bending modes. The variations of the damping with fibre orientations for the first two modes are given in Fig. 11b. The results show that damping is maximum for fibre orientation of about 0° for the first mode, when a maximum of about 45° is observed in the case of the second mode, which is attributed to the distribution of the energies dissipated in the skins and the core of the sandwich materials.



(a)



(b)

FIG. 9 EVALUATION OF THE DAMPING OF THE SANDWICH MATERIALS FOR VARIOUS THICKNESSES OF THE SKIN, IN THE CASE OF (0I/90I)S CROSS-PLY LAMINATE SKINS AND A FOAM DENSITY OF 6060 KG M-3: (a) AS FUNCTION OF SKINS THICKNESSES VALUES, (b) AS FUNCTIONS OF THE RATIO OF CORE AND SKINS THICKNESSES

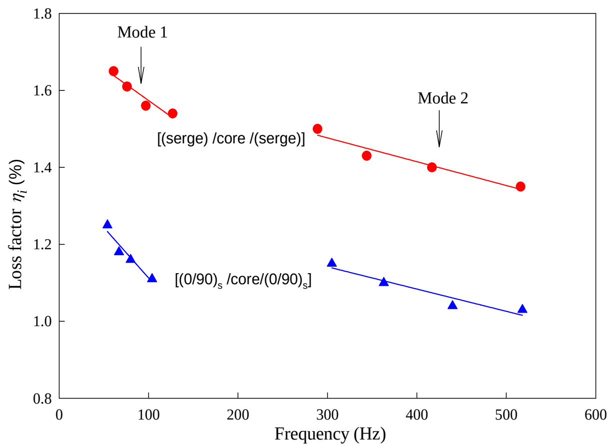
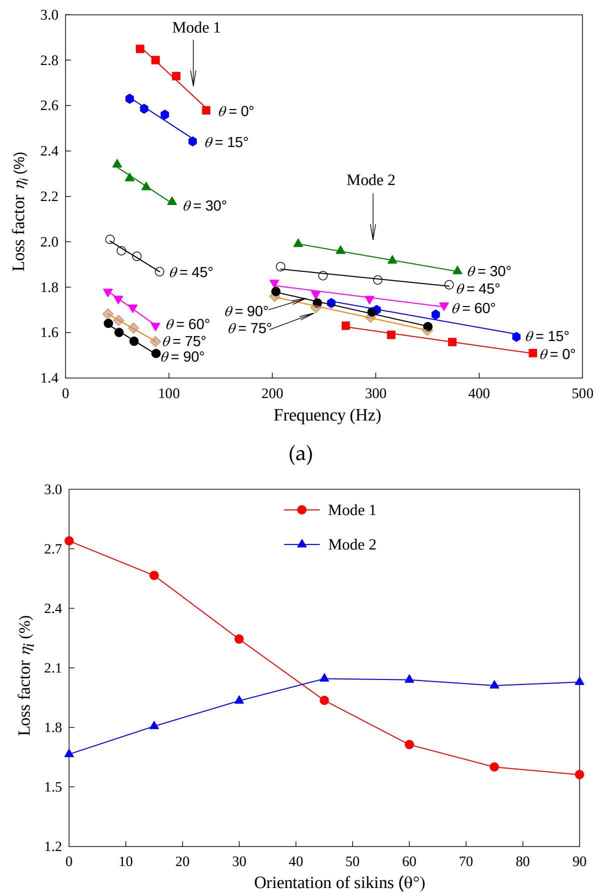


FIG. 10 INFLUENCE OF THE REINFORCEMENT TYPE OF THE SKINS ON THE DAMPING OF SANDWICH MATERIAL, FOR THE FIRST TWO MODES OF BEAMS, IN THE CASE OF CORE WITH FOAM OF 60 KG M-3.

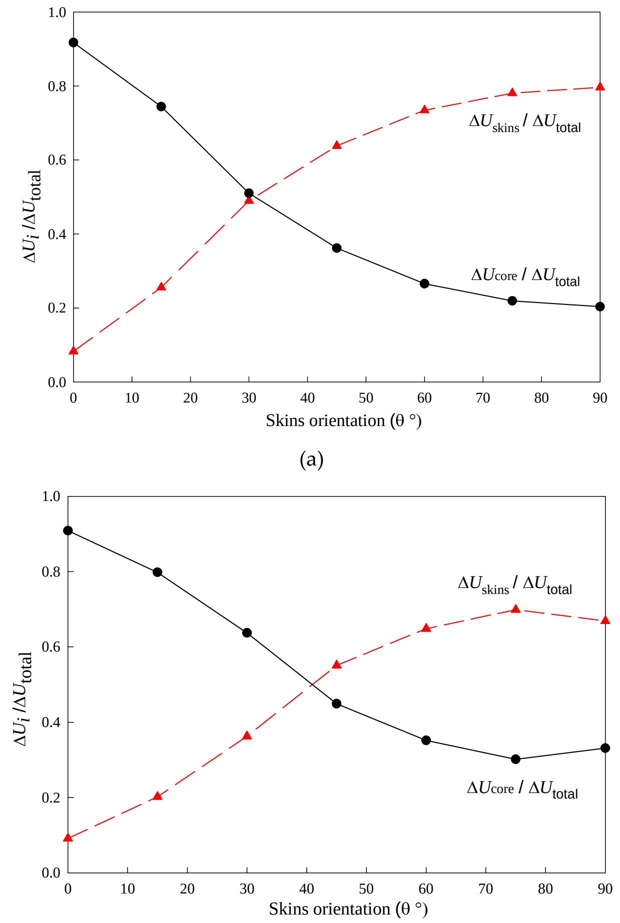


(a)

(b)

FIG. 11 EVALUATION OF THE DAMPING OF THE SANDWICH MATERIALS FOR DIFFERENT SKINS ORIENTATIONS, IN THE CASE OF UNIDIRECTIONAL LAMINATE AND WITH FOAM CORE OF 60 KG M-3: (A) AS A FUNCTION OF THE FREQUENCY, (b)) AS FUNCTIONS OF SKINS ORIENTATIONS.

The relative variations of the energy dissipated in skins and core are reported in Fig. 12. It is observed in the case of the first mode that the energy dissipated in skins increases according to the orientation of skins fibres, while the energy dissipated in the foam core decreases due to the decline of the shear strain in the foam. In fact, when skins fibres are disoriented, the flexural stiffness of the skins decreases, which reduces the stress imposed by skins on the core. This process induces a decrease of the shear energy of the core foam. In the second mode, the variations of the energies dissipated in skins and foam core are similar to those observed in the first mode with a light difference for an orientation of skins fibres of 40°. In this orientation appears the coupling between bending and twisting.



(a)

(b)

FIG. 12 ENERGIES DISSIPATED IN THE CORE AND IN THE SKINS FOR DIFFERENT SKINS ORIENTATIONS, IN THE CASE OF SANDWICH BEAMS OF LENGTHS 350 MM AS FUNCTIONS OF THE ORIENTATION SKINS, (a) FIRST MODE, (b) SECOND MODE.

Conclusions

Modelling of the damping properties of sandwich materials was implemented considering the theory of sandwich plates and using a finite element analysis. The analysis derives the strain energies stored in the material directions of the foam core and in the material directions of the layers of the skins. Further, the energy dissipated by damping in the structure can

be obtained as a function of the strain energies and the damping coefficients associated to the different energies stored in the material directions of the core and the layers of the skins. Next, a parametric study is performed to identify the influence of the characteristics of the constituents on the damping of the sandwich materials. These results obtained show that the main factors influencing the damping of the sandwich materials are: the natural frequencies, the shear modulus of foam, the thickness of the foam, the distribution of the energies stored in the foam and in the skins, the type of reinforcement and the thickness of skins.

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